Energy Savings with Pump Systems
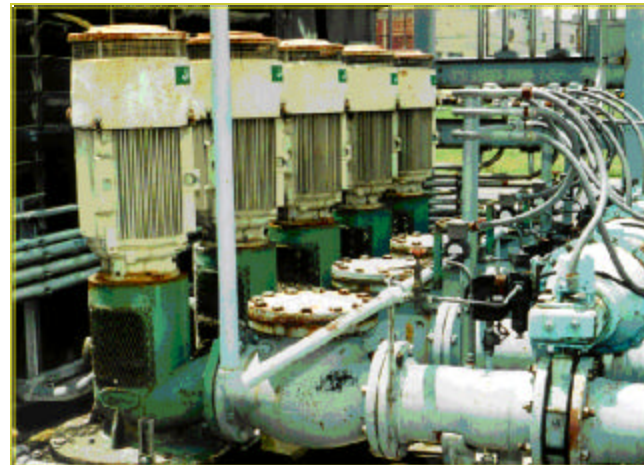
➢How important are pump systems?

➢Some pump system theory

➢Prescreen pump systems: the VITAL FEW and the trivial many

➢Identifying energy savings opportunities

1

Big Picture Perspectives: Industrial Motor Systems
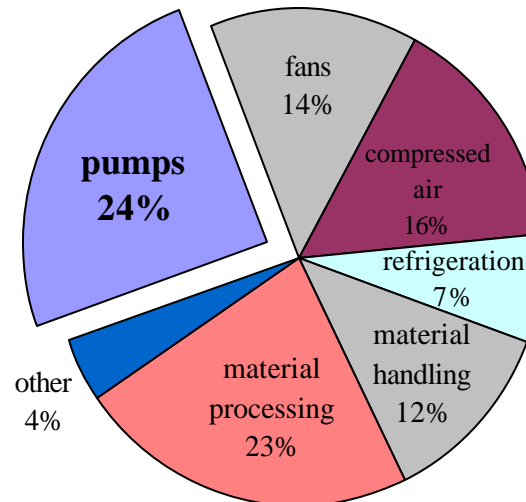
➢Industrial motor systems:

   ➢Are the *single largest electrical end use* category in the American economy

   ➢Account for 25% of U.S. electrical sales

2

Industrial motor energy use by application

Total 2.3 Quadrillion Btu/yr

Pie chart:
- pumps 24%
- fans 14%
- compressed air 16%
- refrigeration 7%
- material handling 12%
- material processing 23%
- other 4%

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Important fundamental relationship

$$\text{Fluid power} = \frac{\text{Flow rate (gpm) x Head (ft) x specific gravity}}{3960} \text{ (hp)}$$

AND

$$\text{Fluid energy} = \text{Fluid power x operating time}$$

Reduce the run time
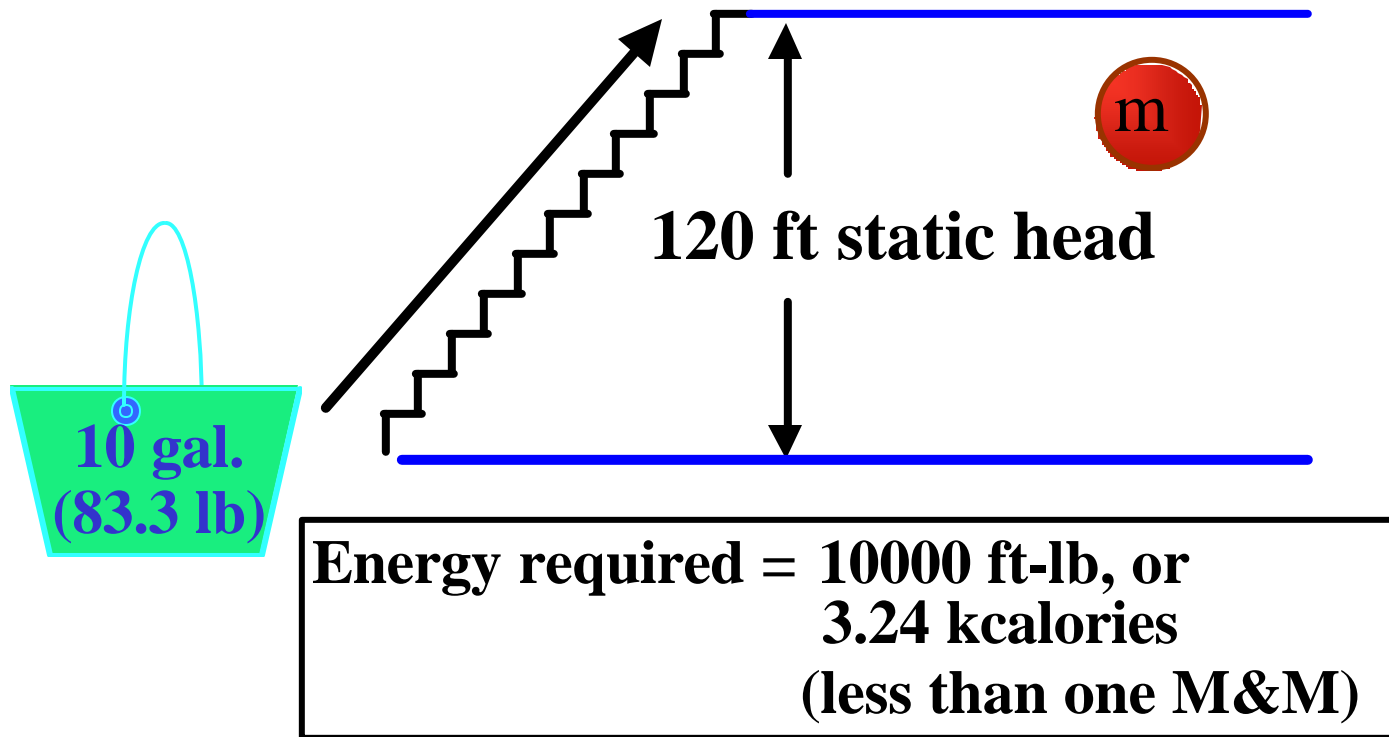**Reduce the flow rate** } **Reduce energy use, cost**
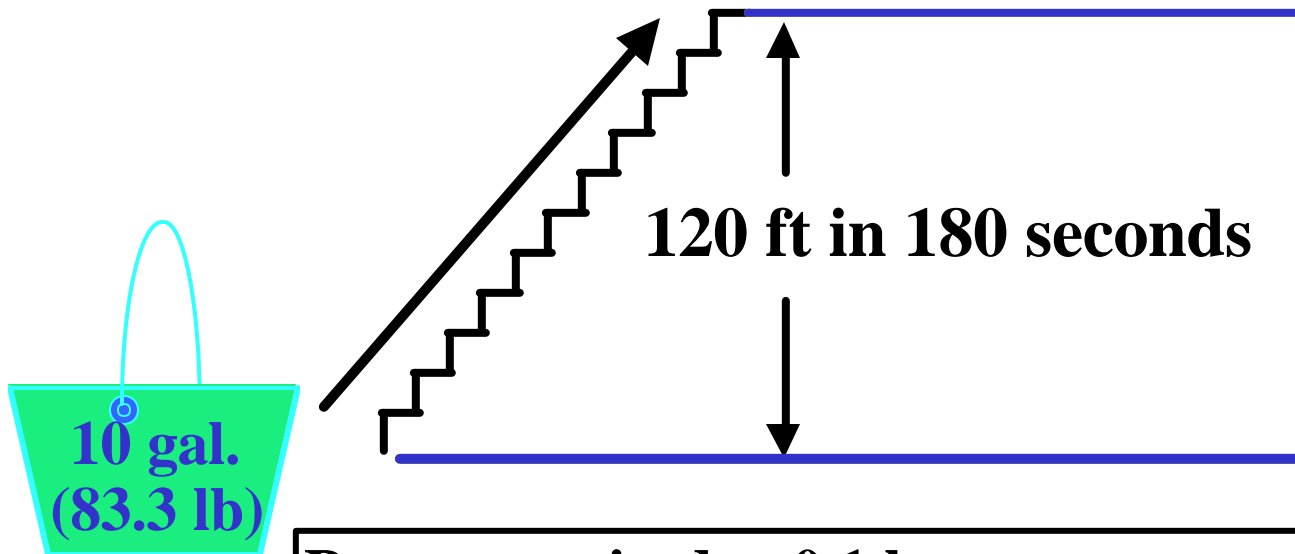**Reduce the head**

4

Ideally, fluid movement energy requirements are proportional to mass and head

120 ft static head

**10 gal.
(83.3 lb)**

m

**Energy required = 10000 ft-lb, or
3.24 kcalories
(less than one M&M)**

Ideal <u>power</u> depends on how fast it is moved

120 ft in 180 seconds
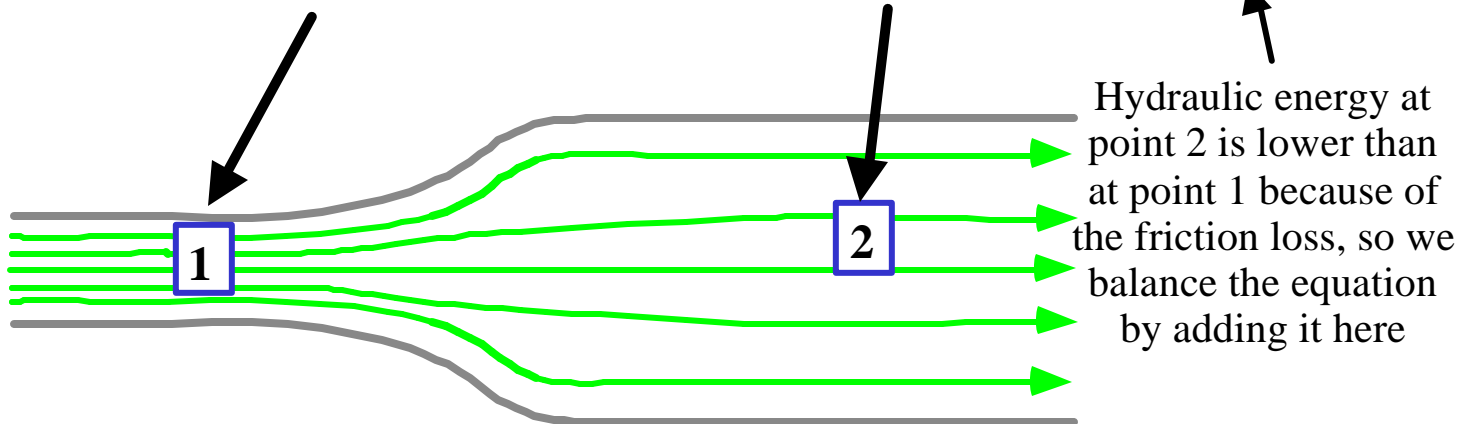
10 gal. (83.3 lb)

Power required = 0.1 horsepower, or 65 kcalories per hour

The Bernoulli equation is slightly modified to account for friction:

$$\left(\frac{V_1^2}{2g} + \frac{P_1}{g} + Z_1 \right) = \left(\frac{V_2^2}{2g} + \frac{P_2}{g} + Z_2 \right) + H_f$$

Hydraulic energy at point 2 is lower than at point 1 because of the friction loss, so we balance the equation by adding it here

Pipe friction loss estimates are usually based on an equation referred to as Darcy-Weisbach

This equation is very useful to understand what parameters influence *frictional* losses in piping:

$$H_f = f \bullet \frac{L}{d} \bullet \frac{V^2}{2g}$$

H_f = pressure drop due to friction (ft)

f = Darcy friction factor

L = pipe length (ft)

d = pipe diameter (ft)

$\dfrac{V^2}{2g}$ = velocity head (ft)

# **Component** friction losses are primarily dependent on experimental data

For pipe components, frictional losses have generally been estimated based on the velocity head.
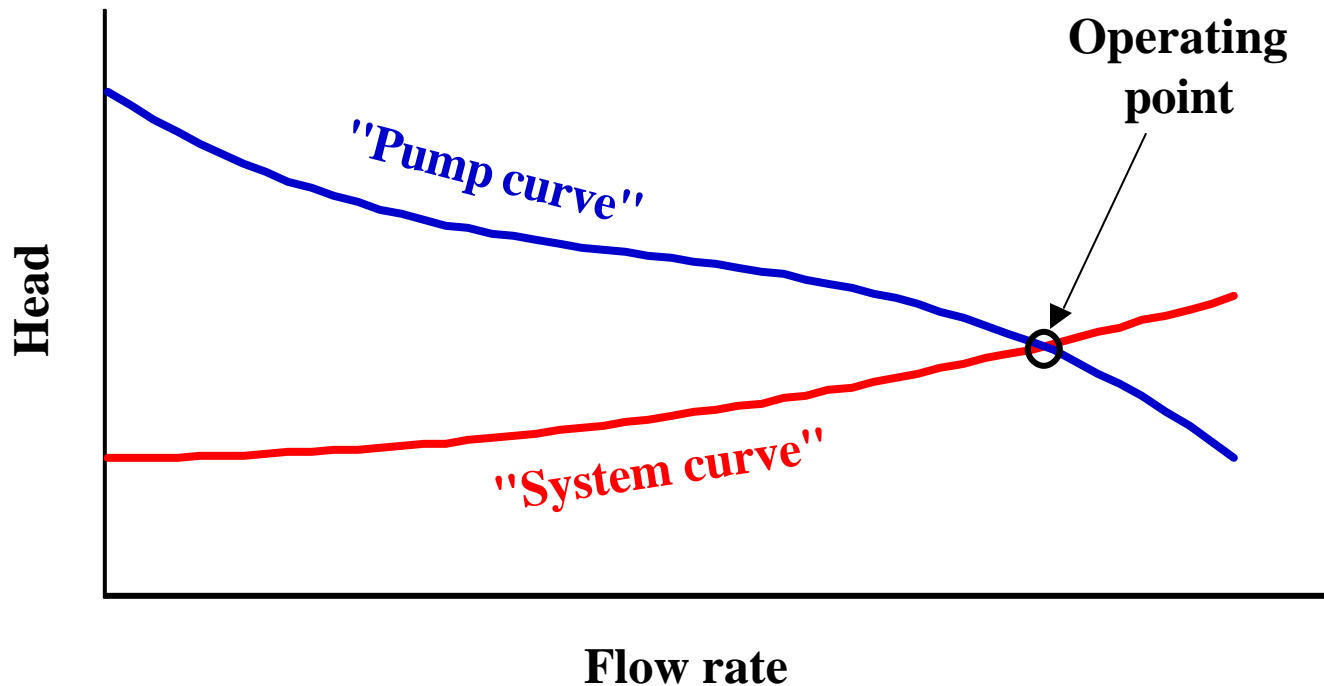
$$H_f = K \cdot \frac{V^2}{2g}$$

K = Loss coefficient

$\dfrac{V^2}{2g}$ = velocity head

K is a function of size, and for valves, the valve type, and valve % open.

System head curves for all frictional, all static, and combined static and frictional systems
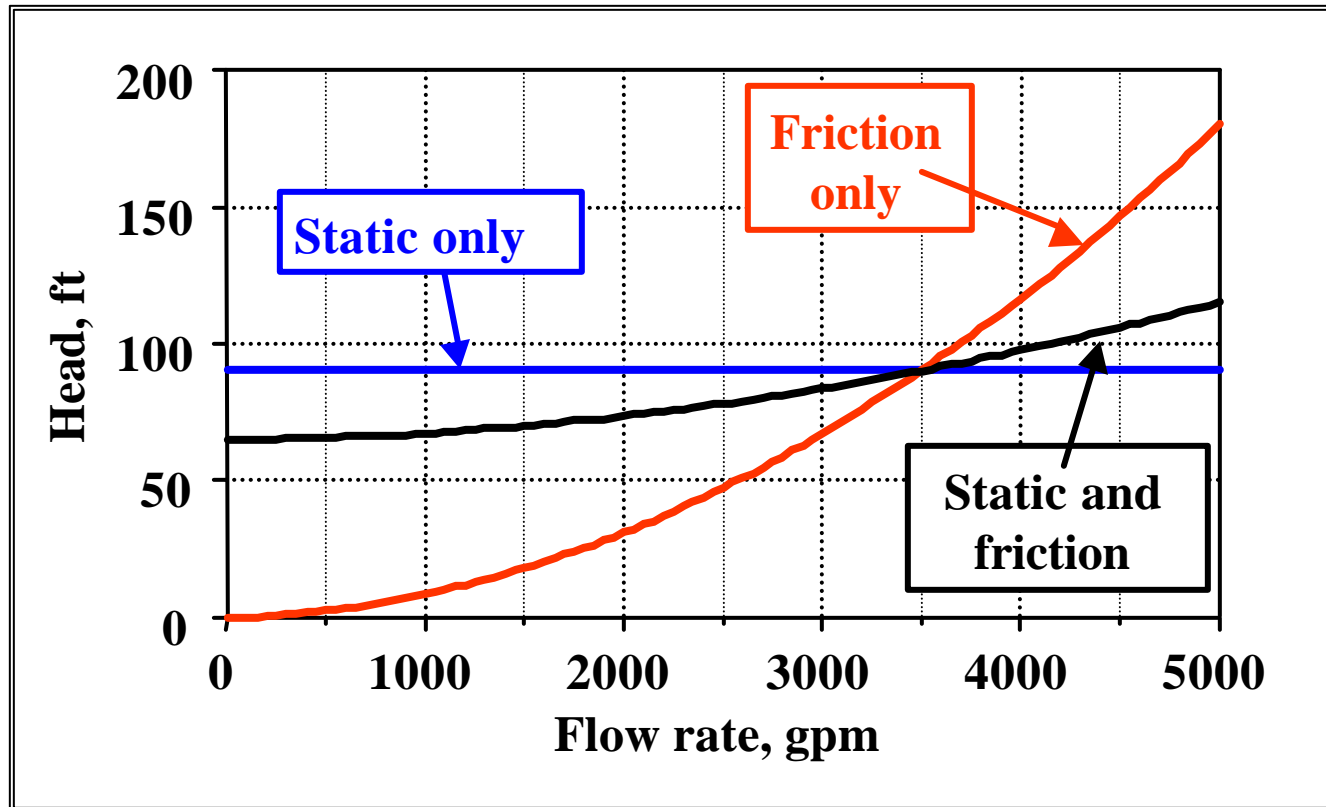
The system operating point is at the intersection of the pump and system head-capacity curves

Operating point

Head

"Pump curve"

"System curve"

Flow rate

Pump affinity laws can be used to predict pump curves for different speeds and impeller diameters

Speed

$$\left(\frac{Q_1}{Q_2}\right) = \left(\frac{N_1}{N_2}\right)^1$$

$$\left(\frac{H_1}{H_2}\right) = \left(\frac{N_1}{N_2}\right)^2$$

$$\left(\frac{P_1}{P_2}\right) = \left(\frac{N_1}{N_2}\right)^3$$
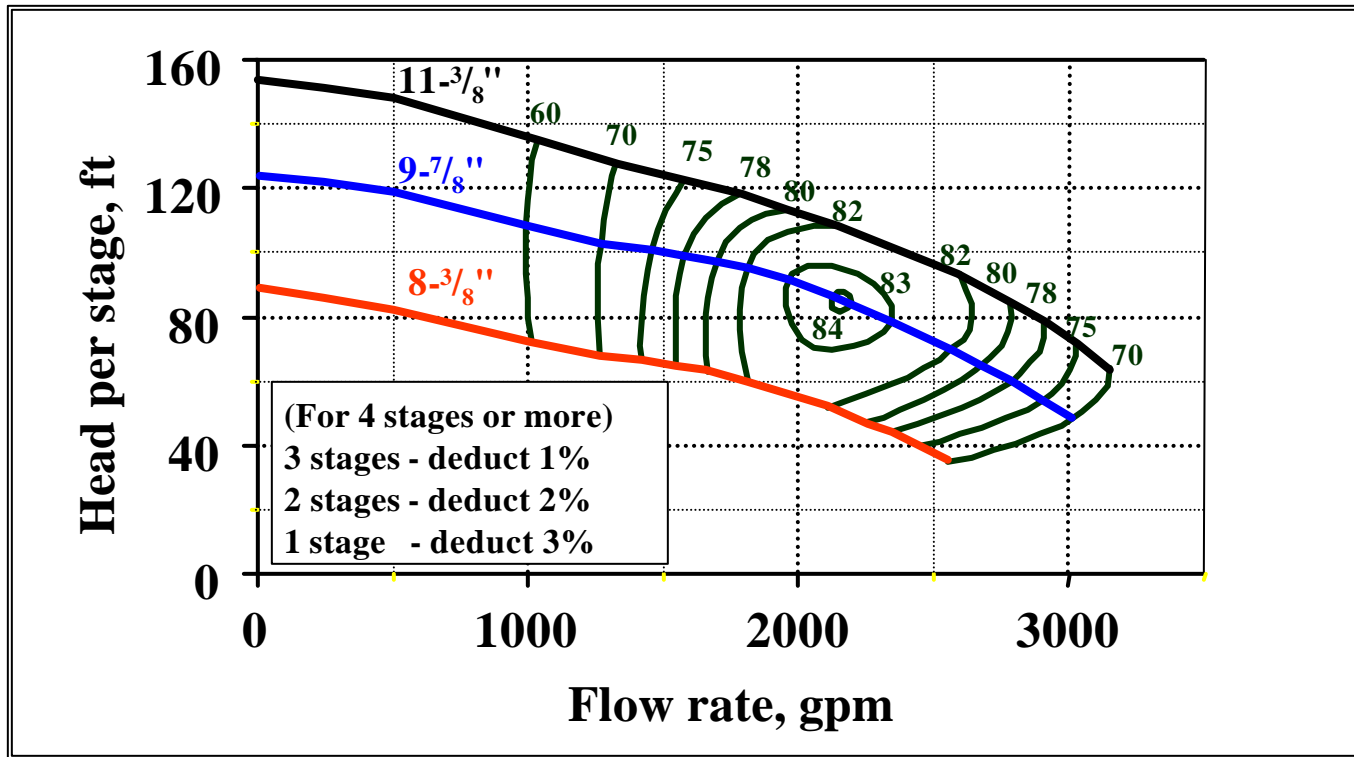
Diameter

$$\left(\frac{Q_1}{Q_2}\right) = \left(\frac{D_1}{D_2}\right)^1$$

$$\left(\frac{H_1}{H_2}\right) = \left(\frac{D_1}{D_2}\right)^2$$

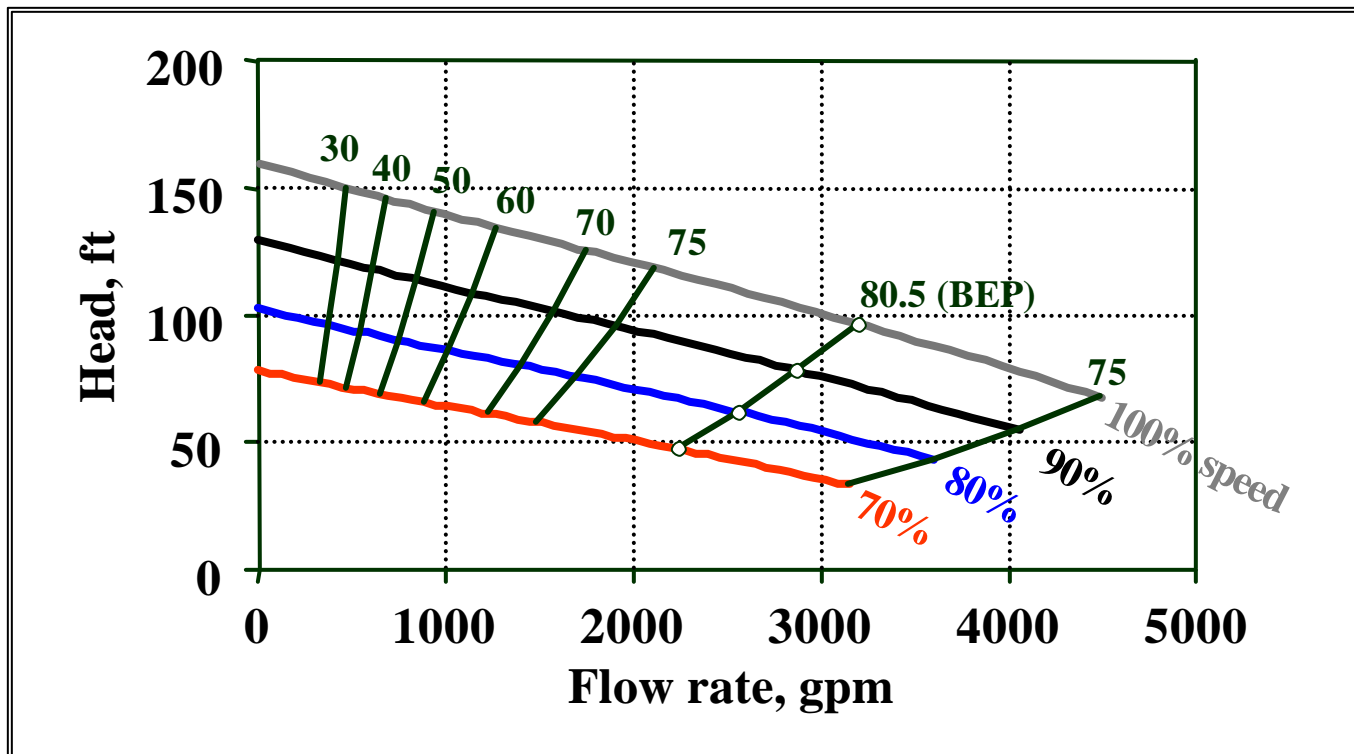$$\left(\frac{P_1}{P_2}\right) = \left(\frac{D_1}{D_2}\right)^3$$

Q = flow rate   H = head   P = power   N = speed   D = diameter

Isoefficiency lines are frequently overlaid onto head-capacity curves for multiple impeller diameters
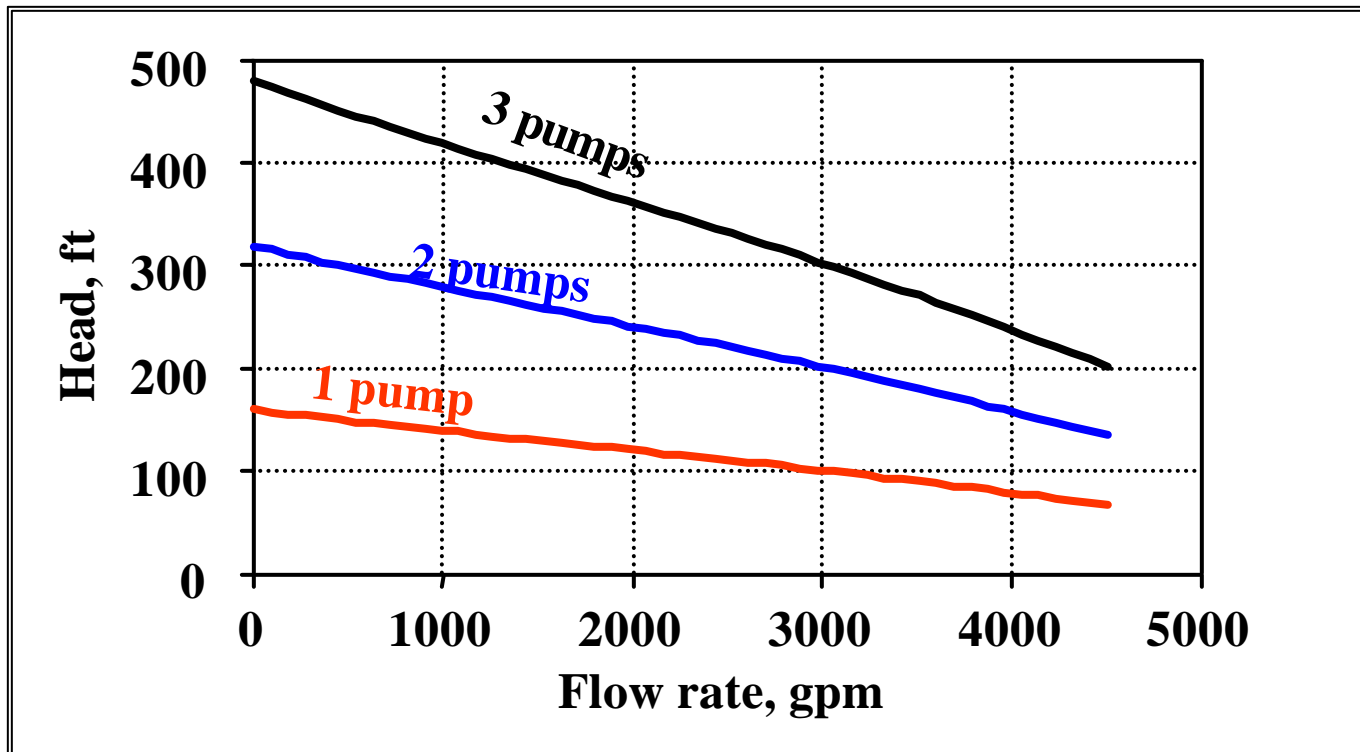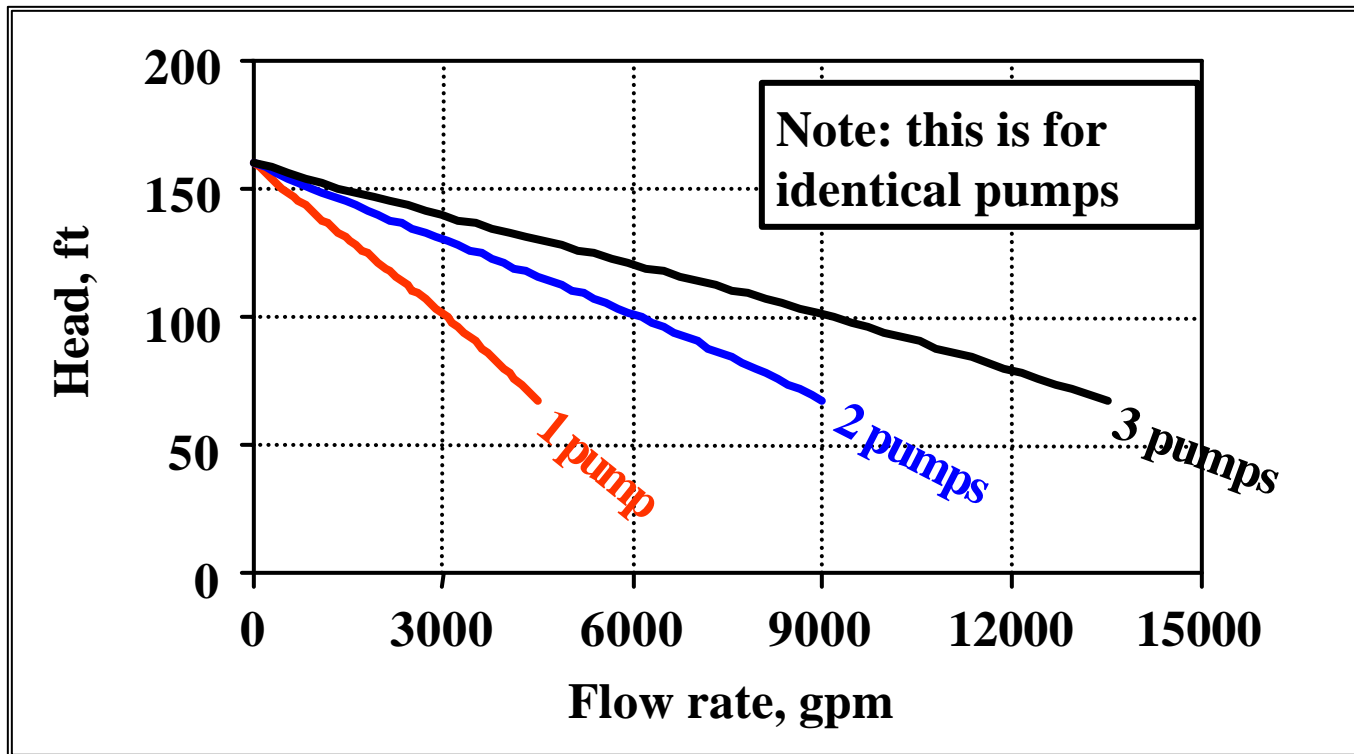
Isoefficiency lines for different shaft speeds

Identical pumps in series; add head at a given flow rate to estimate overall performance

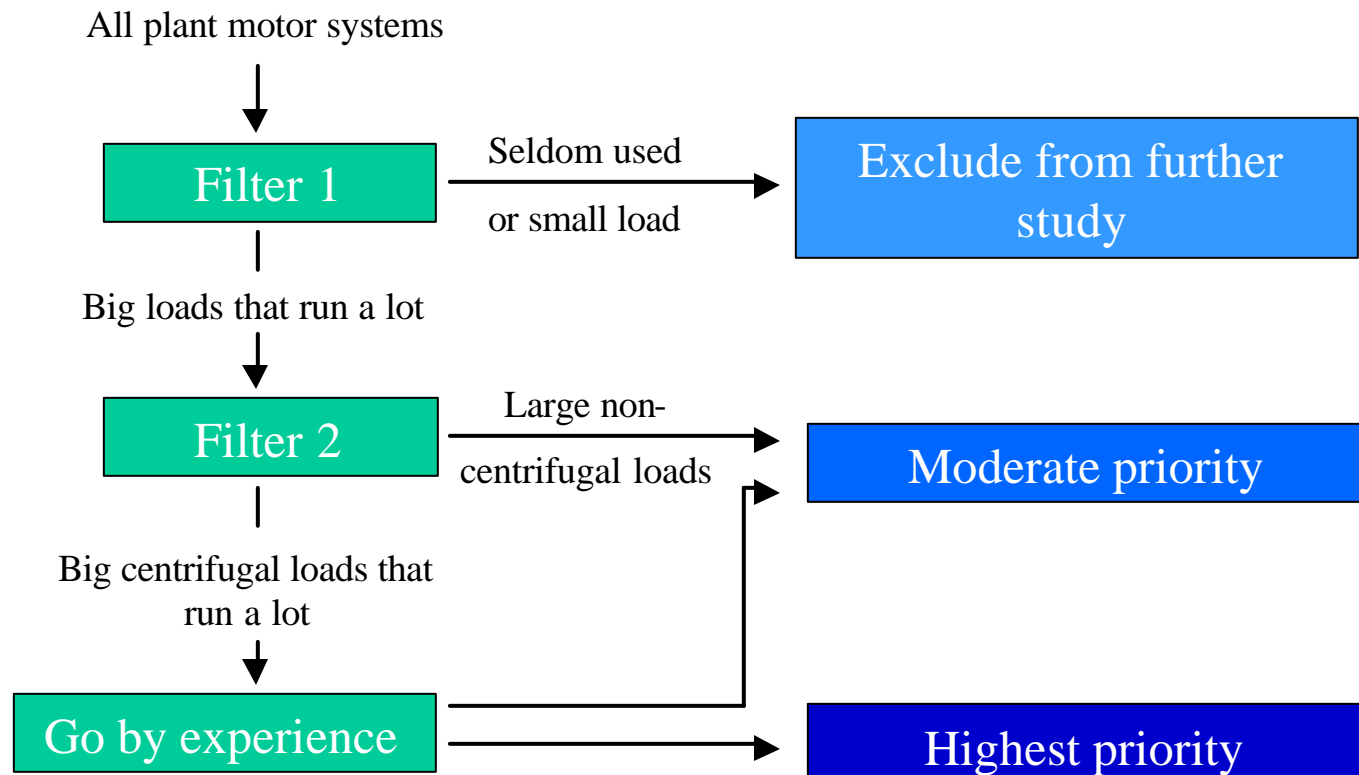# Parallel pumps can help adapt to changing system requirements <u>and</u> provide redundancy

Note: this is for identical pumps

Head, ft (y-axis: 0, 50, 100, 150, 200)

Flow rate, gpm (x-axis: 0, 3000, 6000, 9000, 12000, 15000)

1 pump *2 pumps* *3 pumps*

Select the VITAL FEW and ignore the TRIVIAL MANY

All plant motor systems

↓

| Filter 1 | → Seldom used or small load → | Exclude from further study |

↓ Big loads that run a lot

| Filter 2 | → Large non-centrifugal loads → | Moderate priority |

↓ Big centrifugal loads that run a lot

| Go by experience | → Highest priority |

Symptoms that indicate potential opportunity

Symptom	Likely Reason	Best Solutions
Throttle valve-controlled systems	Oversized pump	Trim Impeller, Smaller Impeller, ASD
Bypass line (partially or completely) open	Oversized pump	Trim Impeller, Smaller Impeller, ASD
Multiple parallel pump system with the same number of pumps always operating	Pump use not monitored or controlled	Install controls
Constant pump operation in a batch environment	Wrong system design	On-off controls
Frequent batch operation in a continuous process	Wrong system design	Match pump capacity with system requirement
Presence of cavitation noise (at pump or elsewhere in the process)	Various	Depends on cause
High maintenance cost (seals, bearings). Talk to operations personnel	Pump operated far away from B.E.P.	Match pump capacity with system requirement

Throttle valve-controlled system

Valve always
partially closed

Centrifugal
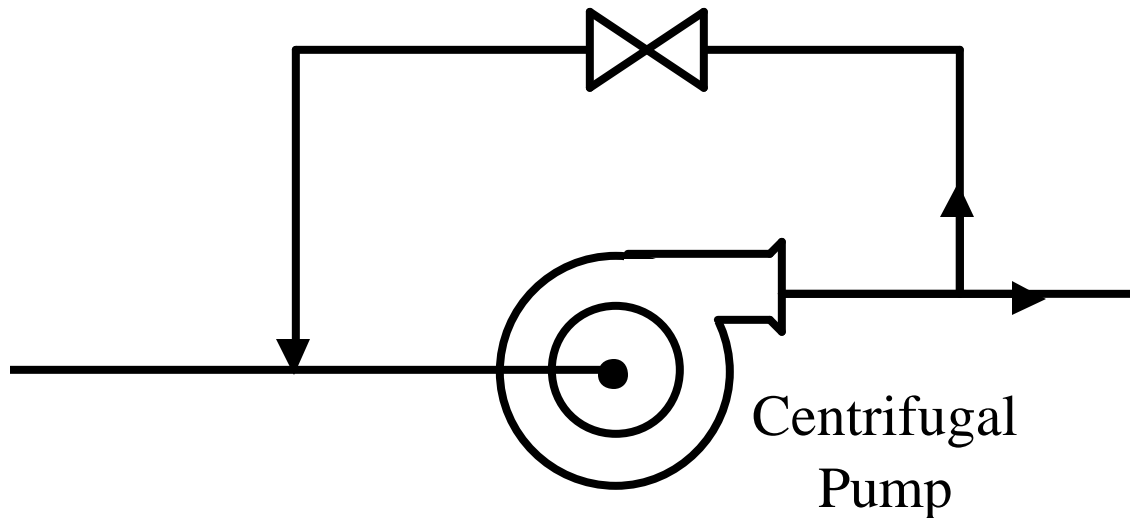Pump

Tank
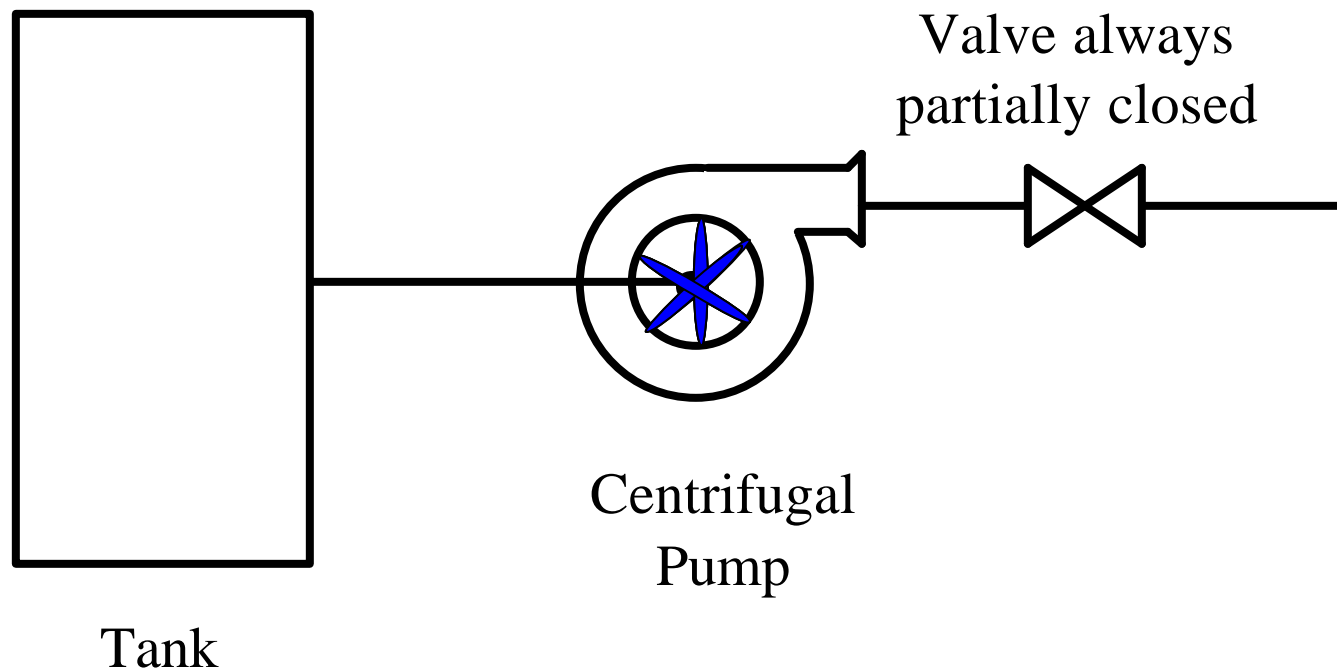
Corrections for oversized pumps include:

➢ Replace the impeller of the existing pump with a smaller impeller

➢ Trim the outside diameter of the existing impeller

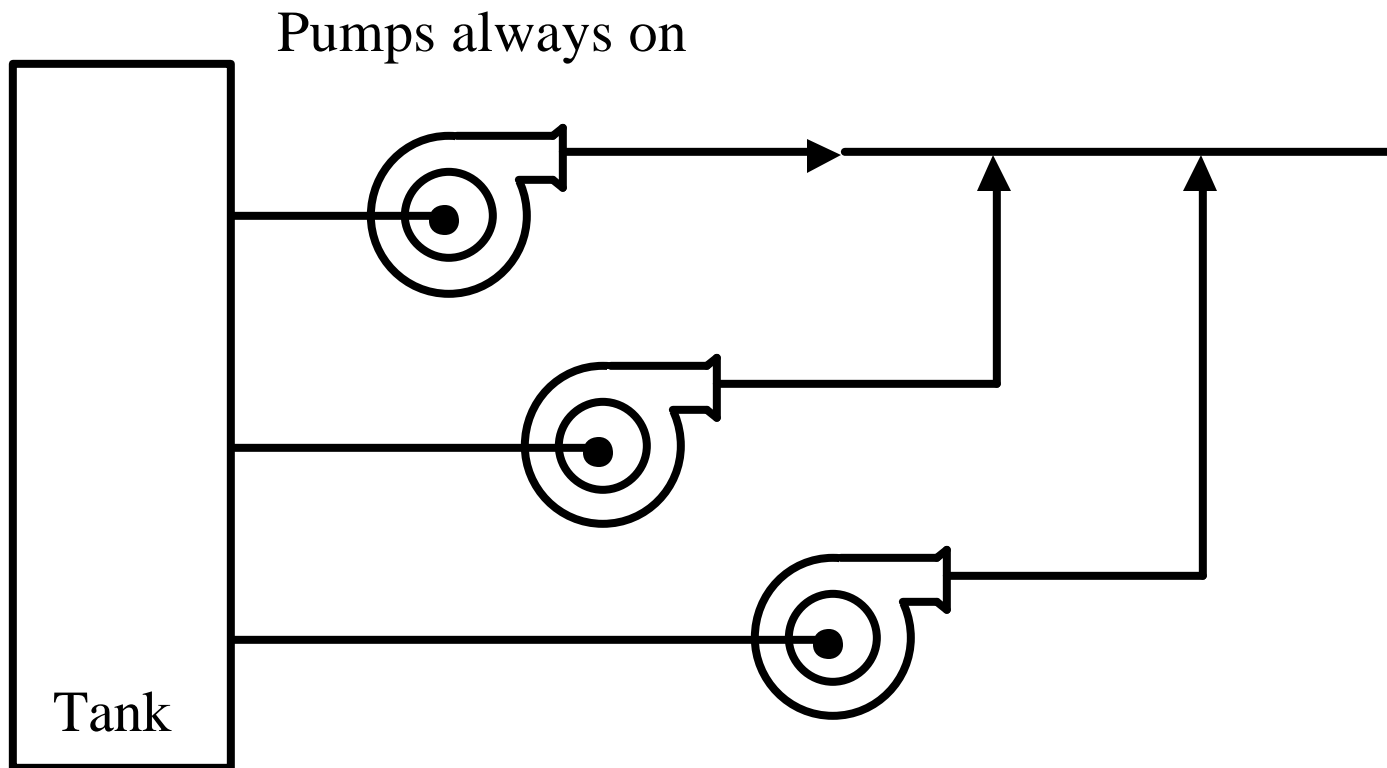➢ Use an adjustable speed drive (ASD) to drive the pump

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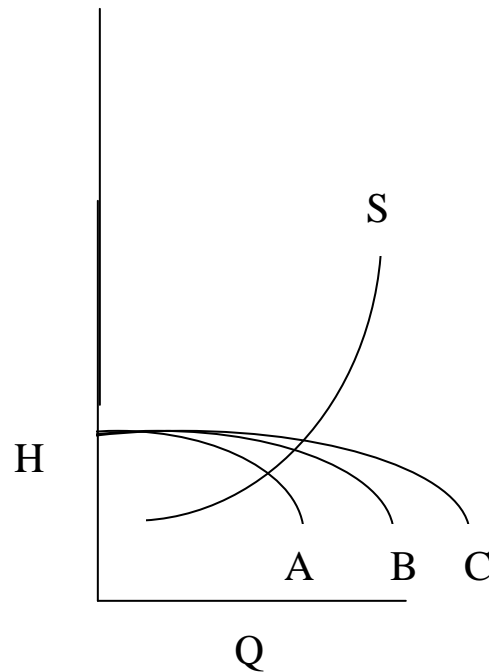# Bypass (recirculation) line partially or completely open

Bypass valve (completely
or partially open)

Centrifugal
Pump

Multiple parallel pump system with same number of pumps always operating
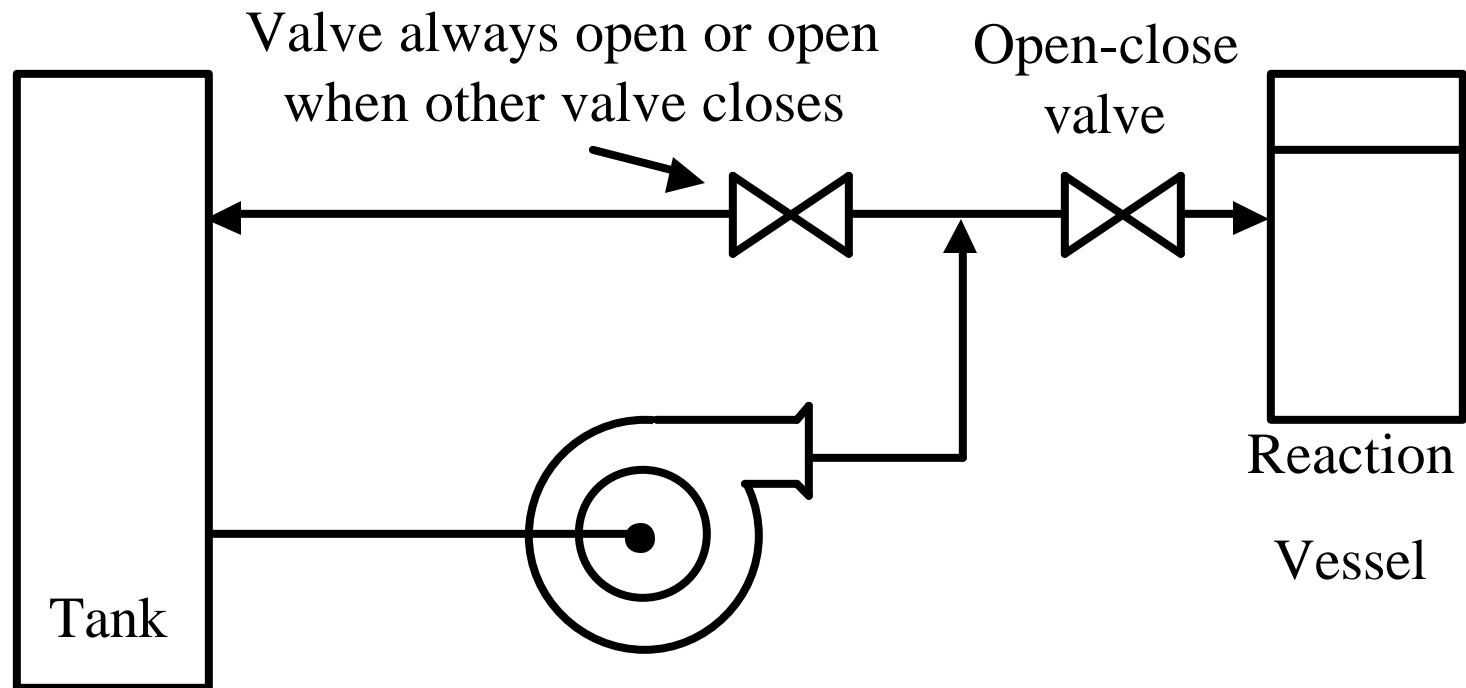
Pumps always on

Tank

Three pumps in parallel

Take a good look at the diagram and you will see that the third pump (C) is intersecting the system curve at just about the same point as the second pump (B).

All of this means that the capacity of three pumps running will not be much greater than that of two pumps running.

S

H

A   B   C

Q

Constant pump operation in a batch environment

Valve always open or open when other valve closes

Open-close valve

Tank

Reaction

Vessel

Frequent cycle batch operation in a continuous process

High level – pump off

Low level – pump on

Tank

Storage

Vessel

Discharge